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(54) Actuating means for a motor vehicle friction clutch has master cylinder operated by electric motor via ball and screw

(57) An actuating means (17) for an hydraulically actuated motor vehicle friction clutch (1), comprises an hydraulic master cylinder (21) and an electric motor (19) connected together to form a sub-assembly, the motor spindle (31) acting through a ball and nut screw drive (27) on the piston (35) of the master cylinder (21). The motor spindle (31), the ball and nut screw drive (27) and the master cylinder are arranged to lie on a common axis. The ball and nut screw drive (27) has an axially displaceable but non-rotatable component (nut 39) and a rotatable but axially fixed component (threaded spindle 37). The axially displaceable component can be acted on by a compensating spring in opposition to the main clutch spring (7). The compensating spring is chosen so that the resulting force exerted on the component reverses its direction at around a position in which the clutch (1) is starting to transmit torque. The actuating means (17) requires only comparatively little space and is able to employ a small electric motor (19).

The actuating means may be used to slip the clutch to avoid driveline vibration.

Fig. 1

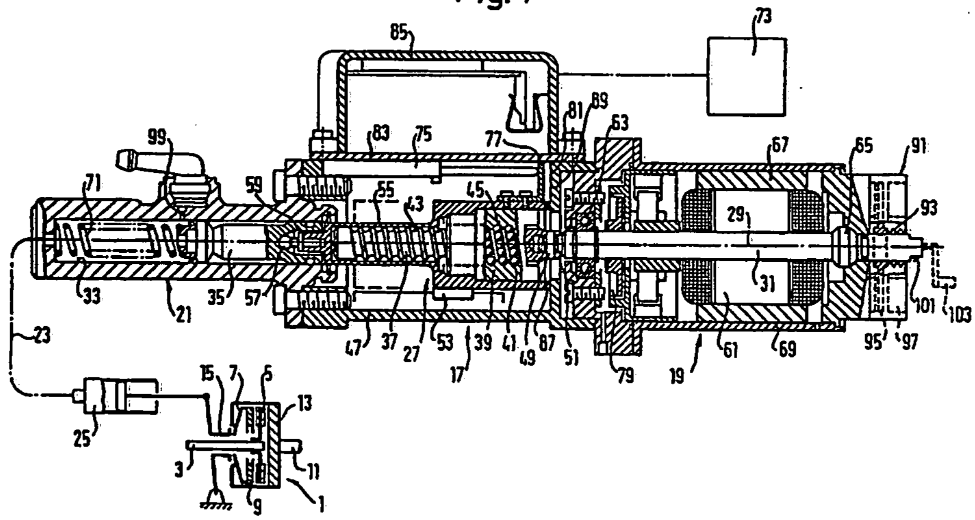

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Fig. 1

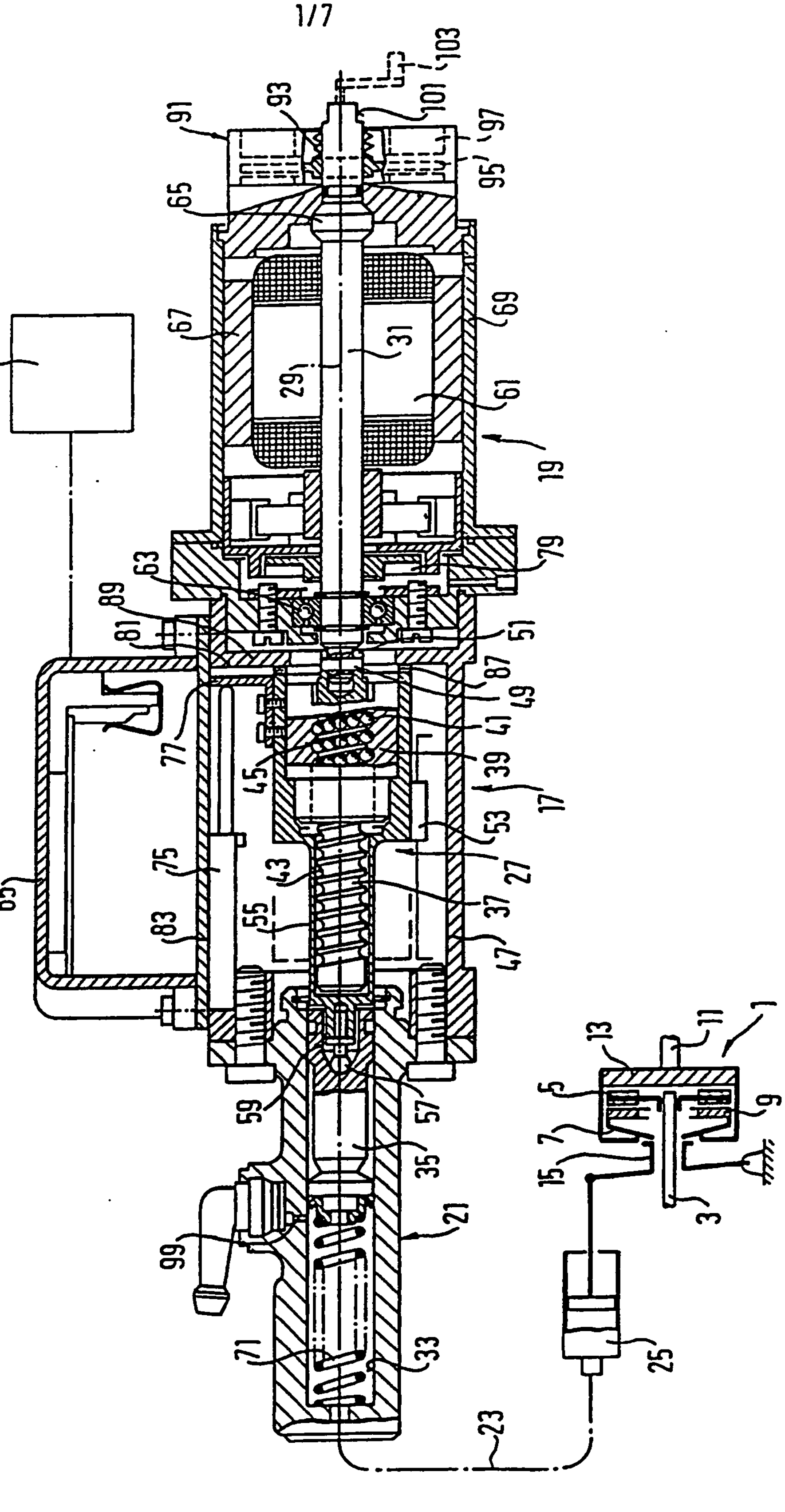

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Fig. 2

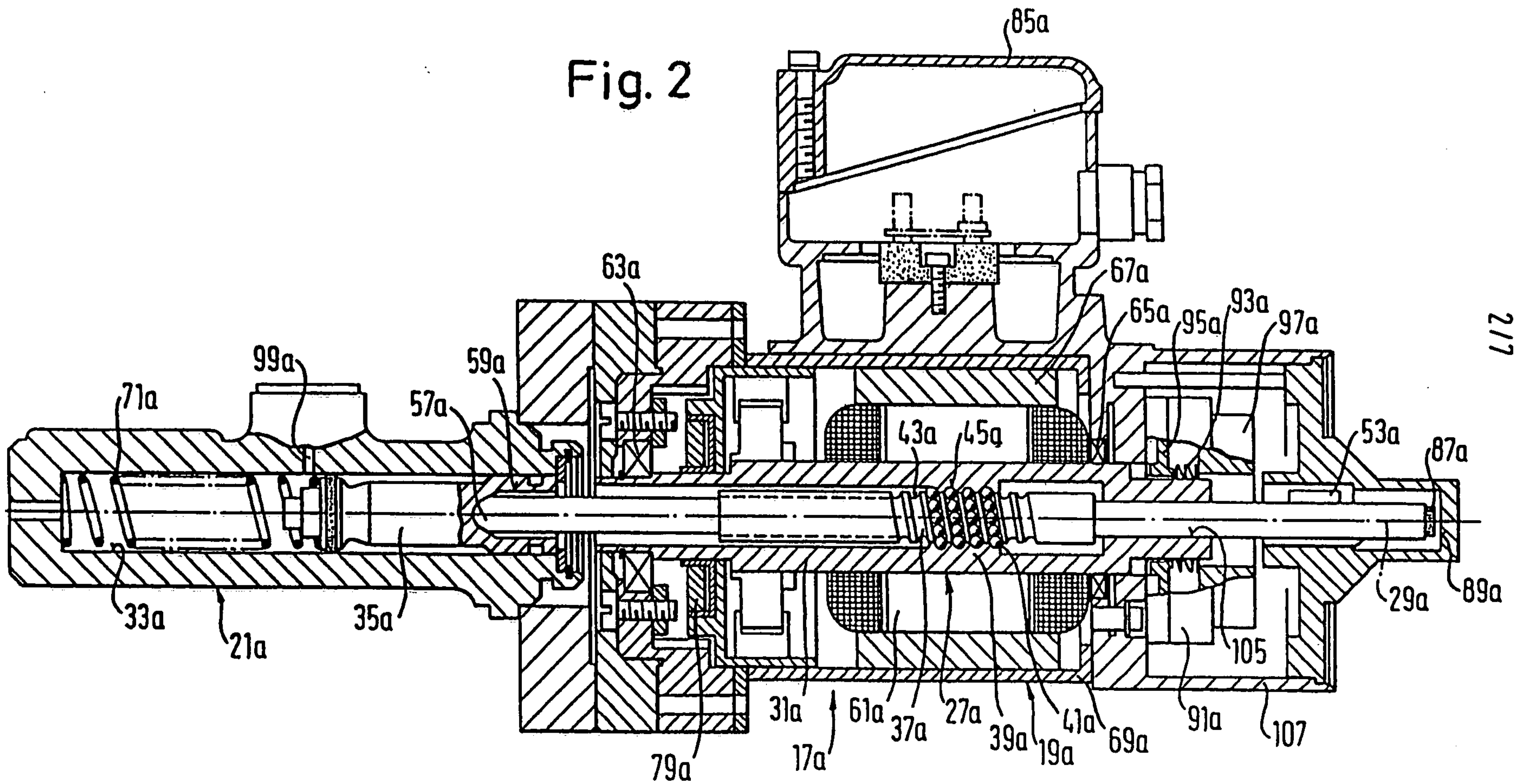
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Fig. 3

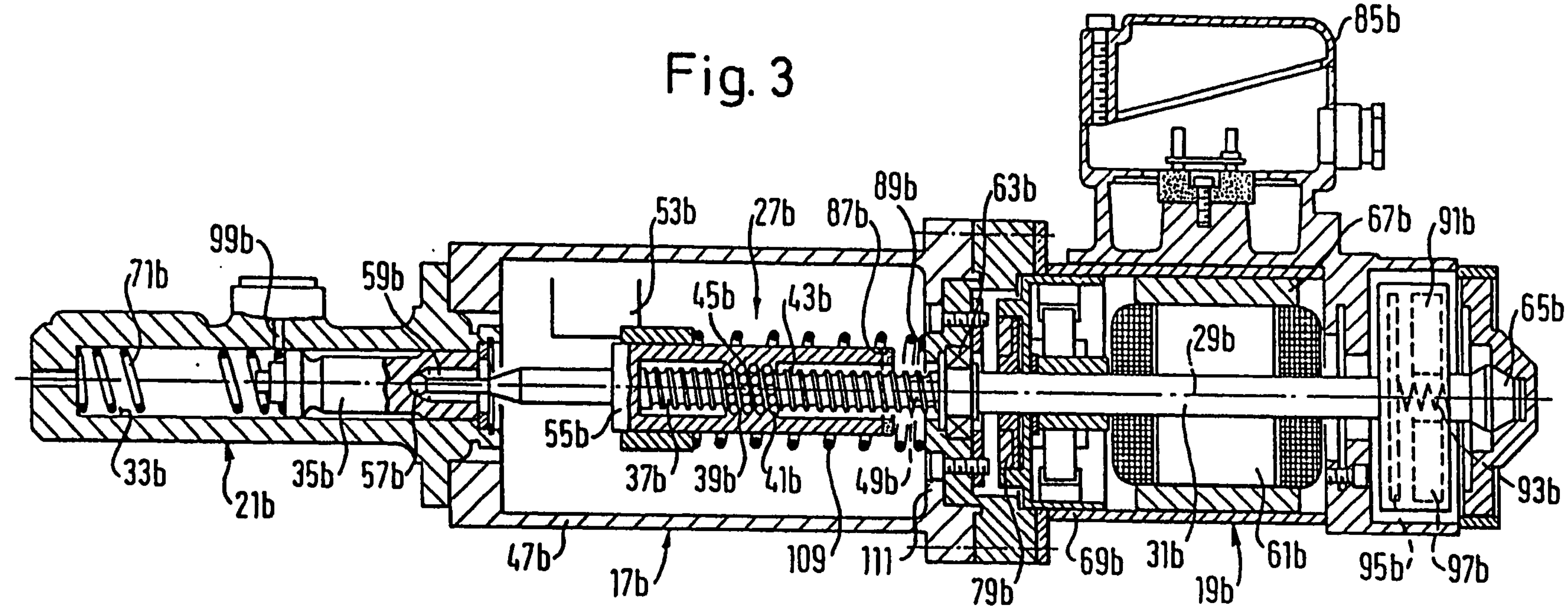

Fig. 4a

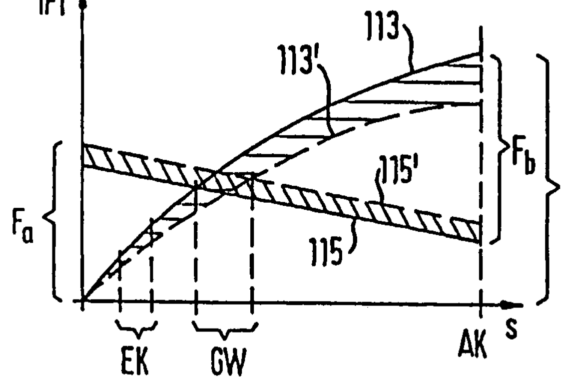

Fig. 4b

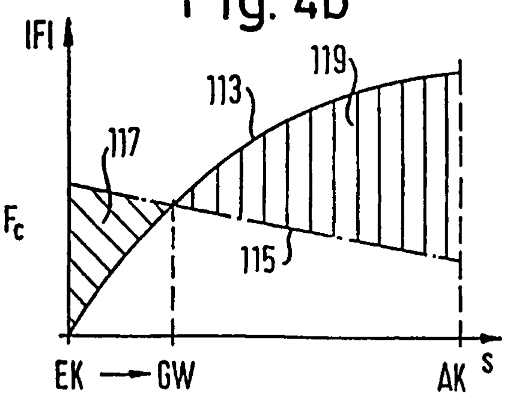

Fig. 4c

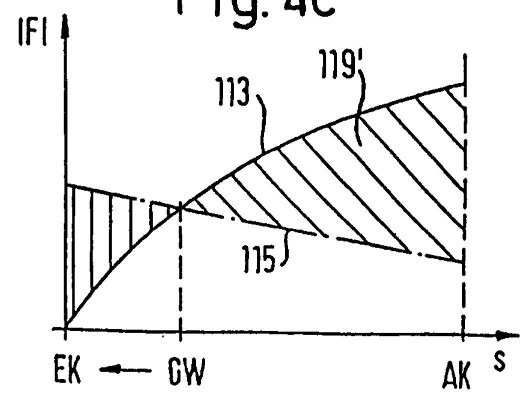

Fig. 6

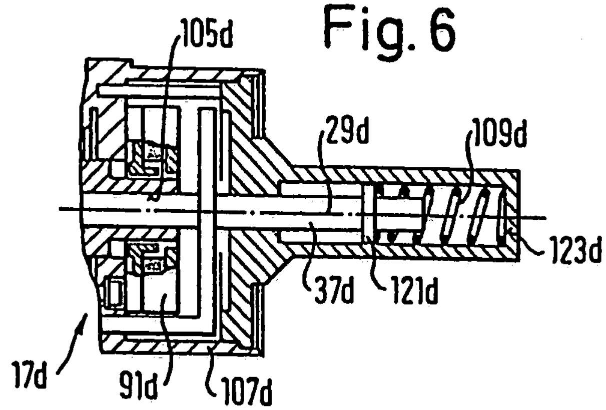
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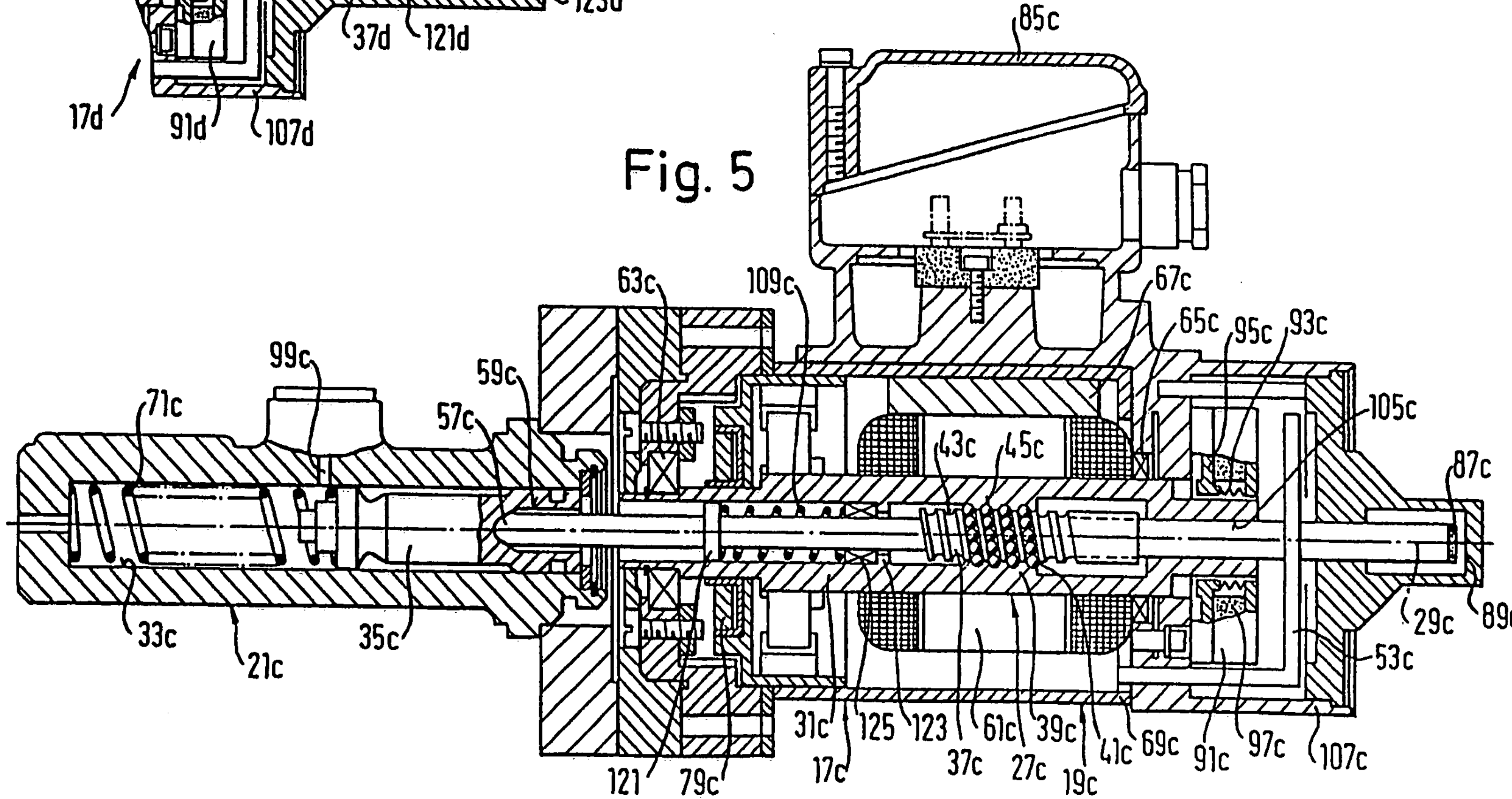
Fig. 5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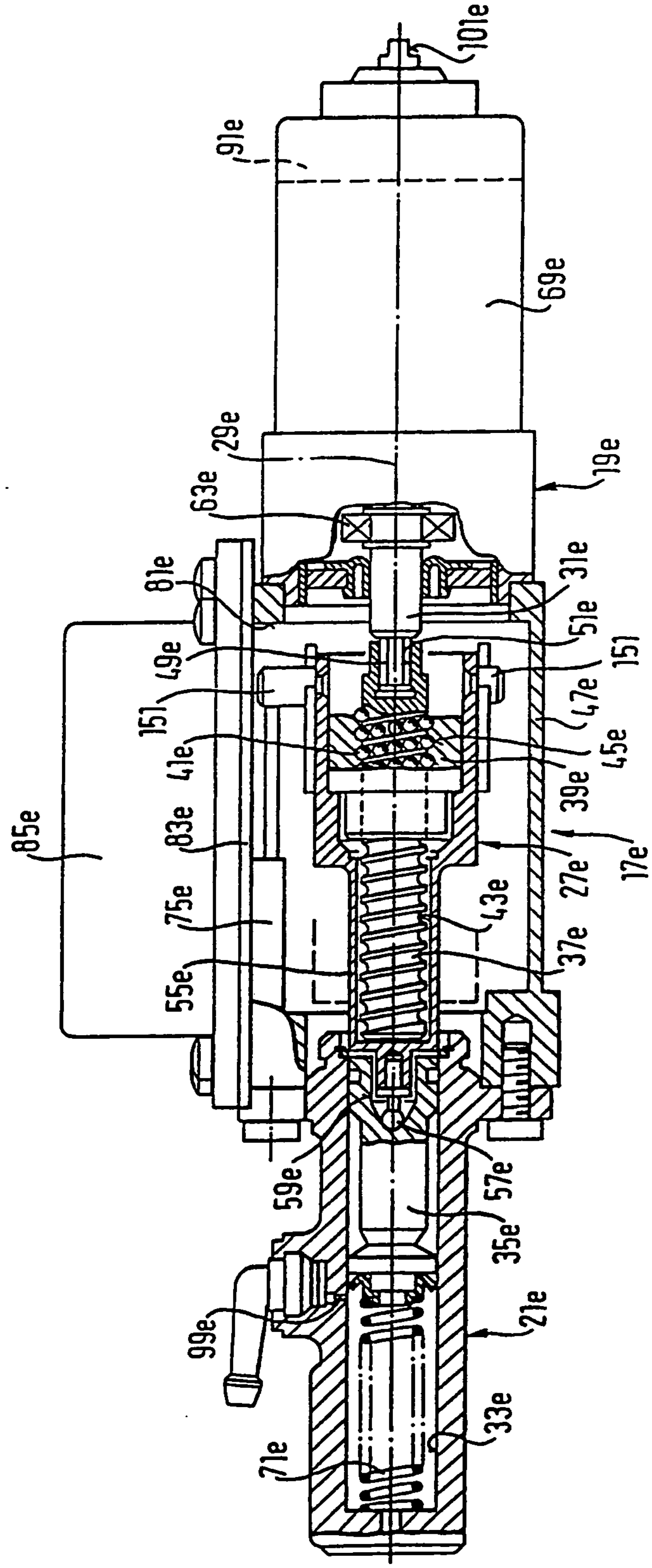
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Fig. 7

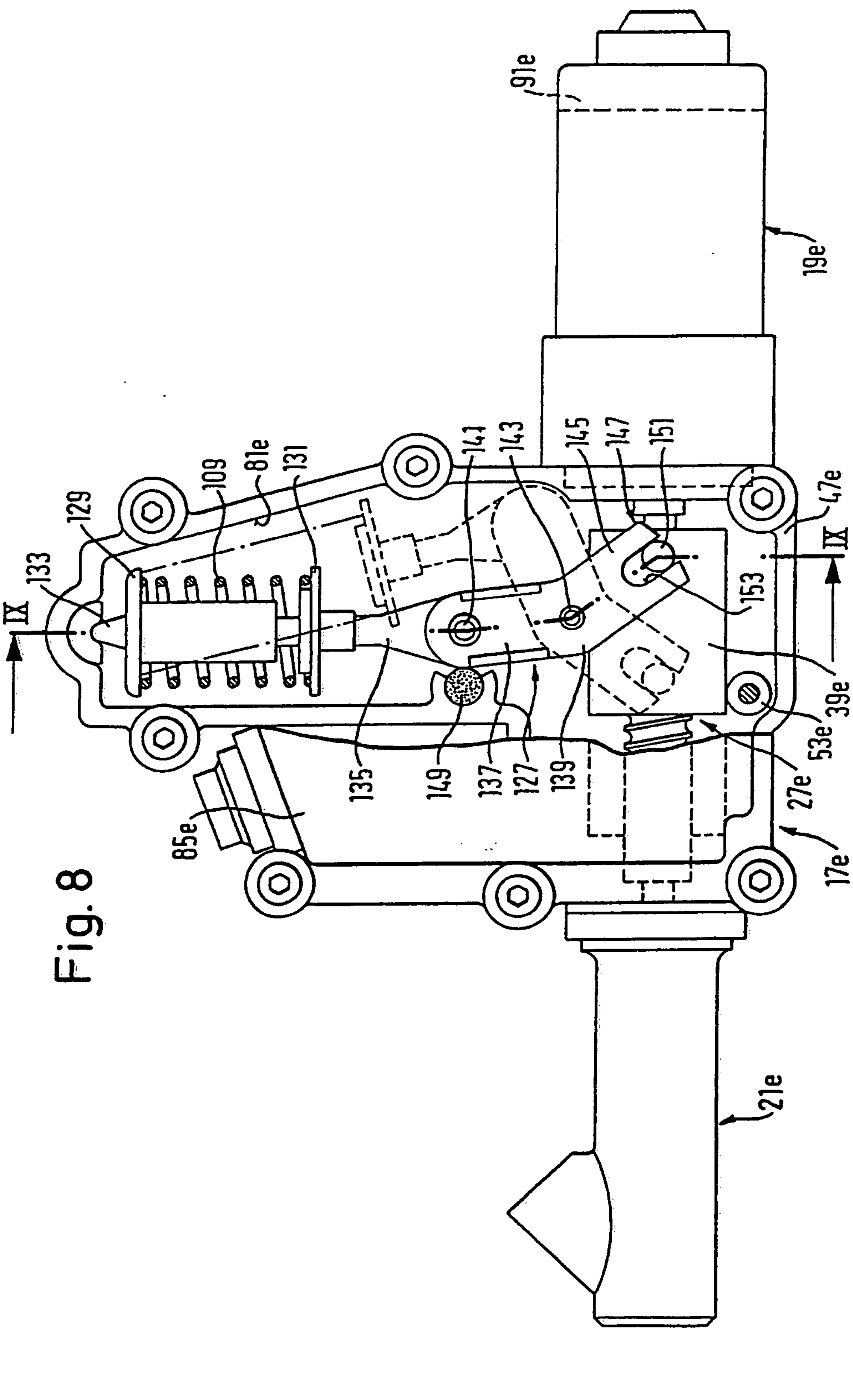
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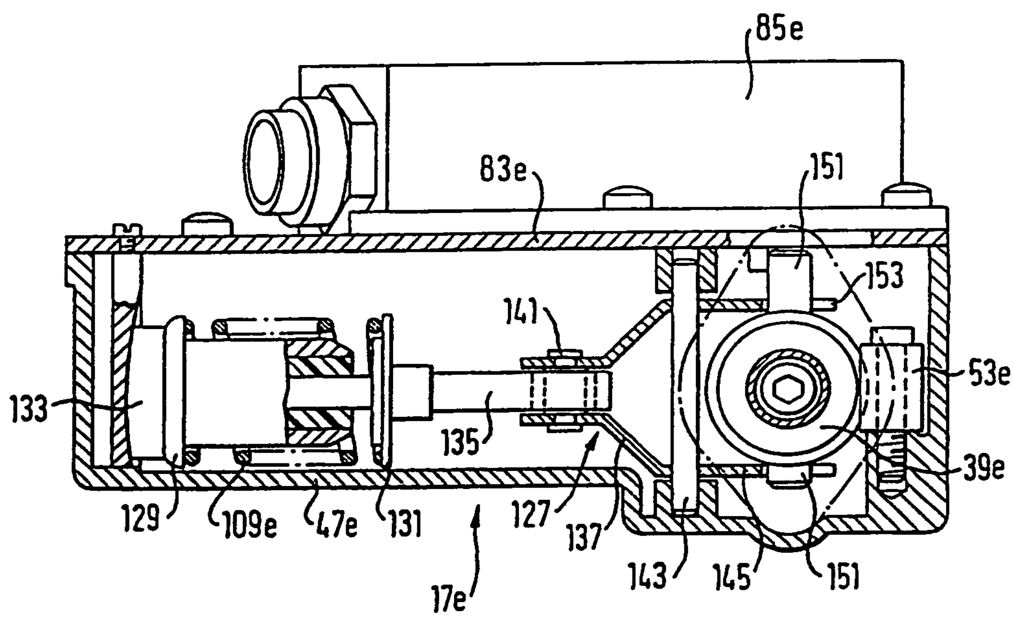
Fig. 8

Fig. 9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ACTUATING MEANS FOR A MOTOR VEHICLE FRICTION CLUTCH

The invention relates to an actuating means for an hydraulically actuated friction clutch of a motor vehicle of the kind comprising an hydraulic master cylinder adapted to actuate an hydraulic slave clutch cylinder and having a piston movable axially in the cylinder, an electric motor connected to the master cylinder to form a subassembly and having a spindle rotating about an axis, and drive means coupling the motor spindle to the piston and converting the rotational movement of the spindle into an axial movement.

Actuating means of the kind set forth are known from (DE-A-39 35 438 and DE-A-39 35 439), and are used instead of a clutch pedal to engage and disengage an orthodox motor vehicle clutch. The actuating means is controlled by an electronic control system which in its turn responds with the aid of sensors to operating parameters of the vehicle, for example engine speed, gearbox input shaft speed and throttle pedal position. The control system engages the clutch automatically on starting and on gear changes in a gearbox of the vehicle. When the vehicle comes to a halt and at the start of a gear-changing sequence the control system disengages the clutch.

From DE-A-33 30 332, and DE-A-34 38 594 and DE-A-36 12 391 it is furthermore known to disengage the clutch slightly in order to reduce rotational vibrations in the drive train. The clutch then transmits the driving torque with a slight degree of slip which eliminates the rotational vibrations superimposed on the driving torque. The rotational vibrations are detected and tuned out by a slip-control

circuit controlling the actuating means. The actuating means may only control the slip, but it can also handle this function in addition to that of automatic actuation of the clutch.

Whereas clutch disengagement must take place comparatively rapidly, a relatively long time is needed for engaging the clutch, in order to prevent as far as possible any thump on engagement. On disengagement the actuating means works against the force of the main clutch spring, and therefore requires a relatively large motor torque and accordingly a relatively large motor.

From DE-A-37 06 849 it is known to actuate the hydraulic slave cylinder of an hydraulic clutch actuating means through an eccentric crank, of which the crank wheel is the form of a worm wheel meshing with a worm mounted on the spindle of an electric motor. A compensating spring bears on the crank wheel and is tensioned by the electric motor as the clutch engages, and then on disengagement it assists the motor in working against the force of the main clutch spring. It has been found that the known actuator takes up a comparatively large amount of space and that, despite the use of the compensating spring, the electric motor must be quite large, so as to provide relatively high reserves of force in order to be able to disengage the clutch in a sufficiently short time.

According to the present invention, in an actuating means of the kind set forth, the motor spindle and the master cylinder are arranged co-axially and the drive means comprises a ball and nut screw drive having a threaded spindle co-axial with the motor spindle and a nut for screwing on the

threaded spindle and in screw-engagement with the threaded spindle through at least one row of balls.

A ball and nut screw drive of this kind allows the rotational movement of the motor to be converted into axial movement with very low friction losses. As the master cylinder, the ball and nut screw drive and the motor spindle are arranged on a common axis, friction-producing transverse forces outside the ball and nut screw drive are avoided, so that a smaller electric motor can be employed. Altogether the result is a more compact arrangement than with the usual actuating means.

In a preferred embodiment the threaded spindle is connected to the motor spindle to rotate with it and the nut abuts axially on the piston of the master cylinder. In this embodiment the ball and nut screw drive is normally located outside the electric motor which is defined by the bearing flange for mounting the motor spindle, which indeed increases the axial structural length but permits the employment of a commercially available ball and nut screw drive. It is then of advantage if the ball and nut screw drive is axially and radially located exclusively on the motor spindle and the master cylinder piston, in order to simplify assembly. This applies in particular where the threaded spindle is connected for rotation with the motor spindle through a dog coupling, whilst the nut abuts loosely against the piston. For supporting the nut on the piston the nut may have a tubular extension enclosing the end of the threaded spindle remote from the motor spindle and being guided on the piston by its end remote from the nut. Such a ball and nut screw drive simply needs to be plugged in on assembly of the electric motor and the master cylinder.

In another embodiment the nut is connected for rotation with the motor spindle, the motor spindle comprising a hollow shaft, and the threaded spindle abuts against the master cylinder piston and extends into the hollow shaft. The nut may be a separate component from the hollow shaft and be mounted on the output end of the hollow shaft for rotation with it. Preferably, however, the nut is arranged within the electric motor; the motor spindle being mounted for rotation in two bearings, and the nut being connected to the motor spindle and being disposed between the bearings. Such an embodiment is extremely compact.

The actuating means usually has an associated travel sensor which generates a signal representing the position of the withdrawal mechanism of the clutch for use in controlling the actuating means. In a preferred embodiment there is provided for this purpose a linear position sensor coupled to the ball and nut screw drive and sensing the position of the component of the ball and nut screw drive which is displaced axially with the master cylinder piston relative to the master cylinder. It has been found to be advantageous if the ball and nut screw drive is arranged in a housing which is connected to the master cylinder and the electric motor to form a sub-assembly also including the linear position sensor. In order to simplify assembly, the housing preferably has an access opening arranged radially to one side of the ball and nut screw drive, the access opening being closed by a wall of a circuit housing containing a control circuit for the motor. This means that a necessary structural component is employed as a closure for the access opening.

The high mechanical efficiency of the ball and nut screw drive may make a compensating spring of the kind

referred to earlier superfluous. If a compensating spring is provided in order to reduce the size of the electric motor it is usually sufficient for the compensating spring to be designed simply for a partial compensation of the opposing force exerted by the main clutch spring. The compensating spring preferably acts on the axially displaceable component of the ball and nut screw drive and the master cylinder piston, over substantially the whole of the range of movement of the clutch in the clutch engaging direction and against an opposing force exerted by the main clutch spring on the displaceable component.

It is particularly advantageous if the compensating spring is chosen so that the force applied to the displaceable component of the ball and nut screw drive with the clutch at a position where it is starting to transmit torque is approximately equal to the opposing force exerted on the displaceable component by the main clutch spring. In this way the force resulting from the force of the compensating spring and the opposing force of the main clutch spring alters in direction in that position in which the clutch is just beginning to transmit torque. Thus, on one side of this reversal position the force of the compensating spring prevails, and on the other side the force of the clutch spring prevails. The arrangement is preferably such that towards the fully engaged position the force of the compensating spring prevails, so that the compensating spring assists the electric motor on disengagement as far as the reversal position. Beyond the reversal position the motor must then work against the resulting spring force, but, in practice this is not substantial when the clutch is disengaged. In the engagement direction the force of the main clutch spring, which prevails over the

compensating spring force, assists the motor in the engagement direction between the disengaged position and the reversal position, so that the initial clearances in the clutch can be very rapidly taken up. Thus, the electric motor only has to work against the resulting spring force in the torque transmitting range of the clutch during the engagement process, where the actuating speed can be comparatively low, so a comparatively small motor can be used.

A further advantage of this partial compensation is that, in contrast to the usual compensating springs, the spring characteristic of the compensating spring only has to be approximately matched to the characteristic of the opposing force of the main clutch spring. In orthodox actuating means, compensating springs matched to the individual situation have to be kept in stock, but with the present invention, the stock can be substantially reduced.

In order to be able to make the electric motor approximately of equal dimensions for both directions of drive it is preferably provided that the magnitude of the difference, in relation to the displaceable component, between the force of the compensating spring and the opposing force with the clutch fully engaged and fully disengaged is approximately equal.

The compensating spring is preferably arranged on the same axis as the threaded spindle and encloses the threaded spindle co-axially. The space taken up by the actuating means can thereby be kept small.

The partial compensation explained above ensures, without additional measures, that when the motor is not

energised, for example on failure of the control system the ball and nut screw drive tends to run towards the reversal position. Preferably therefore additional measures are taken to ensure that the actuating means can be held positively in the fully engaged position of the clutch. The compensating spring may be connected through a two-armed bell-crank lever to the displaceable component of the ball and nut screw drive, the compensating spring engaging a first arm of the lever and a second arm of the lever being pivotally connected to the displaceable component. The bell-crank lever can then adopt an over-dead-centre position, in the region of its position associated with the fully engaged position of the clutch.

Additionally or alternatively, the component of the ball and nut screw drive which is rotatably driven by the electric motor has an associated electromagnetically controlled braking device. This is controlled by the control system which controls the actuating means. In order to be able to keep the brake engaged without current in the rest position, the braking device is advantageously resiliently urged into its braking position and includes a brake-release electromagnet. This only has to be energised during the driving phase of the motor. A disc brake connected to the motor spindle has been found to be particularly suitable.

The two-armed bell-crank lever, because of its arrangement, allows the force-travel characteristic of the compensating spring to match the corresponding characteristic of the main clutch spring. The bell-crank lever, with its over-dead-centre position, can accordingly also be used for embodiments having an

almost complete compensation of the opposing force of the main clutch spring.

As explained previously, motor vehicle friction clutches with actuating means of the kind set forth can be employed for reducing the rotational vibrations in the drive train of the vehicle if clutch slip is introduced in the torque transmission range in dependence on the rotational vibrations detected for example by means of a slip-control circuit. The partial compensation of the opposing force of the main clutch spring makes it possible to control the rotational-vibration-reducing slip using a relatively small electric motor. In order to prevent control shocks it is advantageously provided that the slip-control circuit has a control range which does not overlap the region of the start of torque transmission and thereby the region of the reversal position. This ensures that, despite the hysteresis of the spring characteristic, any lost motion in the force transmission path of the actuating means is always compensated.

Various embodiments of the invention are illustrated by way of example in the accompanying drawings, in which:-

Figure 1 is a diagrammatic illustration of a clutch-actuating installation with an electric motor actuating means in axial longitudinal section;

Figure 2 shows an axial longitudinal section through a modification;

Figure 3 is an axial longitudinal section through an actuating means which can be used in the installation of Figure 1 and has a partial compensating spring;

Figures 4a to 4c are force-travel graphs for explaining the partial compensation;

Figure 5 is an axial longitudinal section through a modification of Figure 3;

Figure 6 is a view of a detail of a modification of Figure 5;

Figure 7 is a partial axial longitudinal section through a force-compensated actuating means which can also be used in the installation of Figure 1;

Figure 8 is a partially broken-away side-view of the actuating means of Figure 7; and

Figure 9 is a section through the actuating means taken along the line IX-IX in Figure 8.

Figure 1 shows diagrammatically an installation for automatically actuating a friction clutch 1 of a motor vehicle. The clutch 1 is of orthodox construction and comprises a clutch plate 5 connected for rotation with a gearbox input shaft 3 and urged by a main clutch spring, for example a diaphragm spring 7, into an engaged position in which it is clamped frictionally between a pressure plate 9 and an opposing pressure plate in the form of a flywheel 13 connected to a crankshaft 11 of the engine of the vehicle. The clutch 1 can be disengaged against the force of the spring 7 by means of a withdrawal mechanism 15.

An electric motor actuating means 17 is provided to actuate the withdrawal mechanism. The means 17 comprises an electric motor 19, for example a direct current commutator motor, which is connected to a hydraulic master cylinder 21 to form a sub-assembly. The master cylinder 21 is connected through an hydraulic pipe 23 to a conventional hydraulic slave cylinder 25 acting on the withdrawal mechanism 15 in order to actuate it. A drive means in the form of a ball and nut screw drive generally indicated at 27 converts the rotational movement of a spindle 31 of the motor 19 about an axis 29 into an axial displacement of a piston 35 of the master cylinder 21, sliding in and sealed in a cylinder bore 33. The ball and nut screw drive 27 has a screw-threaded spindle 37 arranged co-axially with the cylinder bore 33 and with the motor spindle 31, and a nut 39 is arranged to be axially screwed along the spindle 37. The nut 39 is coupled to the spindle 37 through balls 41. The balls 41 are arranged in one or more endless rows in mutually facing helical grooves 43, 45 in the spindle 37 and the nut 39.

The ball and nut screw drive 27 is mounted in a housing 47 to which the motor 19 and the master cylinder 21 are flanged on opposite ends. In the housing 47 the drive 27 is axially and radially guided exclusively by the motor spindle 31 and the master cylinder piston 35. The threaded spindle 37 is rotatably coupled to the motor spindle 31 at its end adjacent the motor, through an axially engageable dog coupling 49 here in the form of a transverse pin which can be inserted in an end slot 51 in the threaded spindle. The nut 39 is guided by rotation-preventing members 53 to be rotationally fixed to, but axially displaceable on, the housing 47. The nut 39 carries a

tubular extension 55 embracing and radially locating the end of the threaded spindle 37 furthest from the motor and this extension is located centrally by means of a guide pin 57 in an opening 59 in the adjacent face of the piston 35.

The motor spindle 31 carries an armature 61 and is mounted in bearings 63, 65 at axially opposite ends of the armature 61, in a motor housing 69 provided with a stator 67. On rotation of the motor spindle 31 the nut 39 is displaced axially. When the nut 39 is driven towards the master cylinder 21 the clutch 1 is disengaged, with the motor 19 working against the force of a return spring 71 of the piston 35, possibly against the force of a return spring, not shown, in the slave cylinder 25 and against the force of the diaphragm spring 7 of the clutch 1. On clutch engagement these springs urge the nut 39 in the opposite direction, and the speed of movement is determined by the energisation of the motor 19. For this the motor 19 is controlled by an electronic control system 73. The electronic control system 73 can be a conventional automatic clutch control system engaging and disengaging the clutch 1 in accordance with the operating parameters of the motor vehicle and in particular of its engine. The control system can also serve other purposes, in particular the deliberate introduction of a slight slip in the transmission of the driving torque in order to eliminate or at least to reduce rotational vibrations in the drive train. Suitable control systems are for example described in the above-mentioned patent applications, DE 33 30 332, DE 34 38 594, DE 36 12 391, DE 39 35 438 and DE 39 35 439.

The control system 73 responds to a number of sensors (not shown), for example speed sensors which represent the engine speed, the gearbox input shaft speed, the gear lever position and the vehicle speed. In order to be able to position exactly the withdrawal mechanism 15 of the clutch 1 the control system 73 acts as a position regulator, to which an actual position signal is supplied from the actuating means 17. The means 17 therefore includes a linear position sensor 75, in the form of a linear potentiometer which is operated through a striker 77 from the axially movable component of the ball and nut screw drive 27, i.e. the nut 39. To increase the accuracy with which the actual position is obtained, an incremental sensor 79 is coupled to the motor spindle 31. The sensor 79 delivers a digital signal proportional to the angular position of the motor spindle 31 and thereby of the threaded spindle 37 which is directly coupled to it.

The analogue linear position sensor 75 is mounted in the housing 47 with the ball and nut screw drive 27. The housing 47 has an opening 81 in its side for inserting these components, the opening 81 being closed by a cover 83. The cover 83 is screwed onto the housing 47 together with a circuit housing 85 and can if necessary be formed by a side wall of the circuit housing 85. The housing 85 contains at least the electronic driving circuit for the motor 19 and possibly also further components of the control system 73.

With the clutch 1 in its fully engaged position the nut 39 abuts against a stop 89 in the housing 47 through a resilient buffer ring 87. With the clutch 1 in its fully disengaged position a stop (not shown) on the clutch limits the travel. As the ball and nut

screw drive 27 has an extremely high mechanical efficiency, which may be more than 90%, the diaphragm spring 7 can displace the drive 27 as well as the motor 19 when the motor is not energised. Thus, an electromagnetic braking device 91 coupled for rotation with the motor spindle 31 and thereby with the threaded spindle 37 holds the drive at least in the fully engaged and fully disengaged positions of the clutch. The braking device 91 is connected to the actuating means 17 to form a unit, and is in the form of a disc brake comprising a brake disc 95, urged into the braking position by a spring 93 and released by an electromagnet 97. The braking device 91 therefore holds the ball and nut screw drive 27 when the electromagnet 97 is not energised. As the operating phases of the motor 19 are short in comparison with its rest phases, the energisation of the magnet 97, which is also controlled from the control system 73, can be restricted to short periods of time, minimising the current used.

A recuperation port 99 connected to a reservoir (not shown) for hydraulic fluid opens into the bore 33 of the master cylinder 21 and is uncovered when the piston 35 is in the position associated with the fully engaged position of the clutch 1, the port being closed off by the piston 35 after a certain degree of lost-motion has been taken-up. Hydraulic fluid flowing into the hydraulic installation from the reservoir in the fully engaged position takes up clearances which could arise as a result of wear, in particular of the friction linings of the clutch plate 5.

The end of the motor spindle 31 remote from the master cylinder 21 is accessible from outside the housing and is provided with keying faces 101 which can

be connected to a crank handle 103 or the like for emergency operation. The ball and nut screw drive 27 can therefore also be operated by hand, for example in the event of failure of the control system 73.

Modifications of the actuating means which can be used in the clutch actuating installation of Figure 1 are shown in the remaining figures. Corresponding reference numerals are used for corresponding parts, with a letter for distinguishing them. In the following examples the components 1 to 15, 23, 25 and 73 are present but are not illustrated.

The actuating means 17a of Figure 2 differs from that of Figure 1 primarily in that the ball and nut screw drive 27a is arranged inside the motor housing 69a between the two bearings 63a and 65a which support the motor spindle 31a. The spindle 31a is in the form of a hollow shaft and is combined as a unit with the nut 39a. The threaded spindle 37a is arranged co-axially inside the hollow spindle 31a and is guided in a central opening 59a in the face of the piston 35a by its supporting end 57a adjacent the master cylinder 21a. The other end of the threaded spindle 37a is guided radially in a bearing opening 105 in the motor spindle 31a and furthermore carries rotation-preventing elements 53a which guide the spindle 37a to be rotationally secured to, but axially movable with respect to a housing 107 which is connected to the motor 19a to form a unit. The housing 107 again carries the casing 85a for the circuit. The stop 87a which determines the fully engaged position of the threaded spindle 27a is provided at the end of the spindle 37a remote from the piston and co-operates with a stop 89a on the housing 107. The linear position sensor 75 is not shown in Figure 2, but could be

present. The nut 39a is formed as part of the motor spindle 31a, and is rotatable, but axially stationary.

It will be understood that the nut 39a can, as in Figure 1, be arranged outside the motor, and the motor spindle which is preferably a hollow shaft, is coupled to the nut in an axial plug-in arrangement in order to simplify assembly of the drive. Keying faces (not shown) may also be provided for emergency operation.

The actuating means 17b in Figure 3 differs from that of Figure 1 in that a compensating spring 109 is associated with the ball and nut screw drive arranged outside the motor 19b. The spring 109 is stressed between the nut 39b, as the axially movable part of the ball and nut screw drive 27b, and a stop face 111 on the housing 47b. The compensating spring 109 exerts on the piston 35b a force opposed to the force exerted by the clutch diaphragm spring through the hydraulic installation. The compensating spring 109 reduces the force which has to be provided by the motor 19b through the ball and nut screw drive 27b on the piston 35b, so that the motor 19b can be smaller. The compensating spring 109 may abut on a stop face on the motor housing 69b rather than on the housing 47b.

Figure 4a shows in a graph the absolute value of the spring force |F| in relation to the spring travel s. Because of friction the spring characteristics are subjected to hysteresis, so that increasing and decreasing loads on the spring follow different paths, in the hatched area in Figure 4a defined between the limiting curves. The line 113 shows the characteristic of the force exerted by the clutch diaphragm spring on the piston 35b during clutch disengagement. The line 113' indicates the characteristic of the diaphragm

spring during clutch engagement. The lines 115 and 115' show the corresponding characteristic of the compensating spring 109 during clutch disengagement and engagement. As shown in Figure 4a, the characteristics of the diaphragm spring and the compensating spring 109 acting on the piston 35b intersect in a region indicated at GW between a position EK in which the clutch is fully engaged and a position AK in which the clutch is fully disengaged. In the region of travel between EK and GW the force of the compensating spring predominates, whereas in the region between GW and AK the force of the clutch diaphragm spring predominates. Thus, the region GW represents a region in which there is a reversal of the direction of the force exerted on the piston 35b by the diaphragm spring and the compensating spring. The compensating spring 109 chosen so that the region of force direction reversal coincides with the position or at least the region in which the clutch is just beginning to transmit a torque.

Figure 4b shows in a simplified representation the relationship during clutch disengagement. In the region 117 the force of the compensating spring 109 prevails, and assists the motor 19b, while it is pushing the piston 35b from the position EK (clutch engaged as shown in Figure 3) to the position GW, in which the clutch is just beginning to transmit torque. This portion of the movement must be passed through rapidly, and this is possible because of the force provided by the compensating spring 109 despite the motor being of relatively small size. Between the positions GW and AK the clutch is already disengaged. The motor 19b is now operating in the region 119 against the predominating force of the diaphragm spring. However, the motor 19b can cope with

the resulting increased load, because the clutch is already disengaged in the region 119.

For clutch engagement, illustrated in Figure 4c the relationships are analogous. In the region 119', moving from the position AK to the position GW, the force of the diaphragm spring predominates, as represented by the characteristic 113'. The diaphragm spring is therefore assisting the motor 19b as far as the position GW. In the region from GW to EK the force of the compensating spring 109 predominates, following the characteristic 115', and so the motor 19b must work against the resulting force. However, as clutch engagement has to take place relatively slowly in order to avoid any possible thump on engagement, this can also be allowed for. Looked at overall, the compensating spring 109, which only partially compensates for the diaphragm spring, allows a reduction in the performance required of the motor 19b. In order to make use of the motor 19b as fully as possible in both directions of movement, the forces F_a and F_b, which result in the positions EK and AK from the difference between the force of the diaphragm spring and the force of the compensating spring, are nearly of the same magnitude. In any case, the differential forces F_a and F_b are smaller than the maximum force F_c which is exerted by the diaphragm spring on the piston 35b.

Figure 5 shows a modification 17c of the actuating means of Figure 2, in which the threaded spindle 37c of the ball and nut screw drive 27c is, similar to Figure 3, loaded by a compensating spring 109c against the force of the diaphragm spring (7 in Fig. 1). The compensating spring 109c acts between an annular shoulder 121 on the threaded spindle 37c and, through

an axial bearing 125, an annular shoulder 123 of the hollow motor spindle 31c which forms the nut 39c. The spring 109c is chosen so that it characteristics correspond to those of Figures 4a to 4c. The force resulting from the compensating spring 109c, the clutch diaphragm spring and the return springs of the hydraulic cylinders acting on the piston 35c of the master cylinder 21c again reverses direction in the region of the position where the clutch is starting to transmit torque. Apart from this the actuating means 17c corresponds to the means 17a of Figure 2, except that the elements 53c which prevent rotation of the threaded spindle 37c abut against a larger diameter on the housing 107c.

Figure 6 shows a detail of a further modification 17d, of the means of Figure 5, in which the compensating spring 109d acts between the shoulder 121d provided on the threaded spindle 37d and a shoulder 123d secured to the housing. This eliminates the axial bearing 125 otherwise necessary to accommodate the rotational movement between the shoulders 121 and 123.

In the embodiments of compensated actuating means in Figures 3, 5 and 6 the compensating spring is arranged co-axially with the motor spindle and produces a force in the direction of that axis. Figures 7 to 9 show an actuating means 17e of which the compensating spring 109e is connected to the nut 39e, as the axially movable component of the ball and nut screw drive 27e, through a lever linkage 127. The construction of the sub-assembly comprising the ball and nut screw drive 27e and the master cylinder 21e corresponds otherwise to the construction of the drive of Figure 1, the compensating spring 109 including the lever linkage 127 being mounted in the housing 47e

which encloses the ball and nut screw drive 27e. The motor spindle 31e is connected to the threaded spindle 37e by a dog coupling 49e, but here in the form of a hexagonal spigot on the motor spindle 31e engaging in a matching socket 51e in the threaded spindle 37e. As shown in particular in Figure 8, the elements which prevent the nut 39e rotating relative to the housing 47e are in the form of rotatably mounted rollers 53e which guide the nut 39e to prevent its rotation but allow axial movement.

As can be seen best in Figures 8 and 9, the compensating spring 109e is 'seated between two supporting members 129, 131, which are guided to slide relative to one another. The member 129 is pivotally supported on the housing 47e through a rocking pivot 133, whilst the member 131 is pivotally connected at 141 to an arm 135 on a first arm 137 of a bell-crank lever 139 (Fig. 8). The bell-crank lever 139 is mounted to pivot on a pin 143 secured to the housing and has its other arm 145 pivotally connected to the nut 39e through a slotted link 147 which allows the movement.

As shown in Figure 8, with the clutch in its fully engaged position, the compensating spring 109e bears against the arm 137 with the bell-crank lever 139 in an over-dead-centre position. In this arrangement the arm 137 abuts against a buffer 149 of resilient material, which is secured to the housing and also forms an end stop for the actuating means in the clutch engaged position. Because of the over-dead-centre position the actuating means 17e is held securely in the clutch engaged position. An electromagnetic brake is therefore not required, but it can be included, as indicated at 91e, in order to be able to hold the means

17e if necessary also in the clutch disengaged position or in intermediate positions.

Figure 8 also shows the bell-crank lever in broken lines in the fully disengaged position. Whereas with the bell-crank lever 139 in the over-dead-centre position illustrated in full lines and corresponding to the clutch engaged position, the spring force acts almost in the direction of the plane of intersection of the pivots 141, 143, in the clutch disengaged position the spring force extends approximately perpendicular to it. The kinematic arrangement of the lever linkage 127 thereby ensures that the torque exerted on the bell-crank lever 139 by the compensating spring 109e increases progressively from the clutch engaged position to the clutch disengaged position, which leads to an alteration of the gradient of the compensating force characteristic 115 illustrated in Figures 4a to 4c. In particular the lever linkage 127 ensures that the compensating force exerted on the piston 35e by the spring 109e is independent of travel, in contrast to the previously described partial compensation.

It will be understood that the compensating spring 109e and the lever linkage 127 can also be matched to one another so that, instead of partial compensation, the force exerted by the spring 109e on the piston 35e in opposition to the force of the clutch diaphragm spring at least nearly fully compensates for the behaviour of the diaphragm spring force over substantially the entire range of movement between the clutch-engaged and clutch-disengaged position.

As shown best in Figure 9, the bell-crank lever 139 is in the form of a fork and embraces the nut 39 on diametrically opposite sides. The slotted link 147 has

trunnions 151 projecting radially from the nut 39e and engaging in slots 153 in the fork at the end of the lever arm 145. One of the trunnions 151 forms at the same time (see Figure 7) a follower stop for the linear position sensor 75e which is likewise mounted in the housing 47e. The actuating means 17e also includes an incremental sensor (not shown) integrated into the electric motor 19e for improving the accuracy by which the position is ascertained.

CLAIMS

1. Actuating means for an hydraulically actuated motor vehicle friction clutch of the kind set forth, in which the motor spindle and the master cylinder are arranged co-axially, and the drive means comprises a ball and nut screw drive having a threaded spindle co-axial with the motor spindle and a nut for screwing on the threaded spindle and in screw-engagement with the threaded spindle through at least one row of balls.

2. Actuating means as claimed in claim 1, in which the threaded spindle is connected to the motor spindle to rotate with it, and the nut abuts axially on the piston of the master cylinder.

3. Actuating means as claimed in claim 2, in which the ball and nut screw drive is axially and radially located only on the motor spindle and the master cylinder piston.

4. Actuating means as claimed in claim 2 or claim 3, in which the threaded spindle is connected for rotation with the motor spindle through a dog coupling.

5. Actuating means as claimed in any of claims 2 to 4, in which the nut abuts loosely against the piston.

6. Actuating means as claimed in claim 5, in which the nut carries a tubular extension enclosing the end of the threaded spindle remote from the motor spindle and being guided on the piston by its end remote from the nut.

7. Actuating means as claimed in claim 1, in which the nut is connected for rotation with the motor

spindle, the motor spindle comprising a hollow shaft, and the threaded spindle abuts against the master cylinder piston and extends into the hollow shaft.

8. Actuating means as claimed in claim 7, in which the nut is a separate component from the hollow shaft, and is mounted on the output end of the hollow shaft for rotation with it.

9. Actuating means as claimed in claim 7, in which the motor spindle is mounted for rotation in two bearings and the nut is connected to the motor spindle and is disposed between the bearings.

10. Actuating means as claimed in any preceding claim, in which the ball and nut screw drive is arranged in a housing connected to the master cylinder and the motor to form a sub-assembly also including a linear position sensor connected to the ball and nut screw drive.

11. Actuating means as claimed in claim 10, in which the housing has an access opening arranged radially to one side of the ball and nut screw drive, the access opening being closed off by a wall of a circuit housing containing a control circuit for the motor.

12. Actuating means as claimed in any preceding claim, in which the sub-assembly comprising the master cylinder and the motor has a compensating spring which acts on the displaceable component of the ball and nut screw drive and the master cylinder piston over substantially the entire range of movement of the clutch in the clutch engaging direction and against an opposing force exerted by a main clutch spring on the displaceable component.

13. Actuating means as claimed in claim 12, in which the compensating spring chosen so that the force which it applies to the displaceable component of the ball and nut screw drive with the clutch at a position where it is starting to transmit torque is approximately equal to the opposing force exerted on the displaceable component by the main clutch spring.

14. Actuating means as claimed in claim 12 or claim 13, in which the force applied to the displaceable component of the ball and nut screw drive by the compensating spring with the clutch at a position where it is transmitting torque is greater than the opposing force exerted on the displaceable component by the main clutch spring and in which, with the clutch at a position where it transmits substantially no torque, is smaller than this opposing force.

15. Actuating means as claimed in claim 14, in which the magnitude of the difference, in relation to the displaceable component, between the force of the compensating spring and the opposing force with the clutch fully engaged and fully disengaged is approximately equal.

16. Actuating means as claimed in any of claims 12 to 15, in which the compensating spring is arranged on the same axis as the threaded spindle and encloses the threaded spindle co-axially.

17. Actuating means as claimed in any of claims 12 to 15, in which the compensating spring is connected to the displaceable component through a two-armed bell-crank lever, the compensating spring engaging a first arm of the lever and a second arm of the lever being pivotally connected to the displaceable component.

18. Actuating means as claimed in claim 17, in which the bell-crank lever takes up an over-dead-centre position in the region of its position associated with the fully engaged position of the clutch.

19. Actuating means as claimed in any preceding claim, in which the component of the ball and nut screw drive which is rotatably driven from the electric motor has an associated electromagnetically controlled braking device.

20. Actuating means as claimed in claim 19, in which the braking device is resiliently urged into its braking position and includes a brake-release electromagnet.

21. Actuating means as claimed in claim 20, in which the braking device comprising a disc brake connected to the motor spindle.

22. Actuating means as claimed in any of claims 12 to 21, in which in order to reduce rotational vibrations in the drive train of the motor vehicle with the clutch, the electric motor is controllable by a slip-control circuit dependent on rotational vibrations to apply a slip to the clutch and the slip-control circuit has a control region which does not overlap the region of the start of torque transmission.

23. Actuating means for an hydraulically actuated motor vehicle friction clutch of the kind set forth substantially as described herein with reference to and as illustrated in Figure 1 of the accompanying drawings.

24. Actuating means for an hydraulically actuated motor vehicle friction clutch of the kind set forth

substantially as described herein with reference to and as illustrated in Figure 2 of the accompanying drawings.

25. Actuating means for an hydraulically actuated motor vehicle friction clutch of the kind set forth substantially as described herein with reference to and as illustrated in Figure 3 of the accompanying drawings.

26. Actuating means for an hydraulically actuated motor vehicle friction clutch of the kind set forth substantially as described herein with reference to and as illustrated in Figure 5 of the accompanying drawings.

27. Actuating means for an hydraulically actuated motor vehicle friction clutch of the kind set forth substantially as described herein with reference to and as illustrated in Figure 6 of the accompanying drawings.

28. Actuating means for an hydraulically actuated motor vehicle friction clutch of the kind set forth substantially as described herein with reference to and as illustrated in Figures 7 to 9 of the accompanying drawings.

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Relevant Technical Fields

(i) UK Cl (Ed.M) F2L (LAX), F2F (FFA)

(ii) Int Cl (Ed.5) B60T 13/74

Databases (see below)
(i) UK Patent Office collections of GB, EP, WO and US patent specifications.

(ii) ONLINE DATABASES: WPI, EDOC

Search Examiner
JIM CALVERT

Date of completion of Search
25 JULY 1994

Documents considered relevant following a search in respect of Claims :-
1-28

Categories of documents

X: Document indicating lack of novelty or of inventive step.

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Category	Identity of document and relevant passages	Relevant to claim(s)
X	WO 89/03783 A1 (AUTOMOTIVE PRODUCTS) Whole document	1-4, 7
X	WO 89/03782 A1 (AUTOMOTIVE PRODUCTS) Whole document	1, 10
X	US 4858436 (ROLTRA) See the Figure	1
X	US 4812723 (HONDA) See eg. Figure 1	1

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